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# Centrifugal Compressor Performance Deviations with Various Refrigerants, Impeller Sizes and Shaft Speeds

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## ABSTRACT

Under the pressure of the Montreal Protocol originally written in 1987 (UNEP, 1987), the current popularly used HCFC and HFC refrigerants for comfortable air conditioning, such as low pressure refrigerant R123, will have to be phased out in the near future. This forces companies in centrifugal chiller industry to redesign and retest new centrifugal compressors which can accommodate alternative refrigerants. Keeping the present flow path designs is an economic design process for the alternative refrigerants, and is well adapted by companies in this industry. In addition extensively employ the flow similitude theory is also an effective way to minimize the cost of lab tests. However employing similitude in compressible flow does not forecast the content in which compressors meet the uncontrollable situations such as choke and surge. Conclusions can only be drawn from experimental data. (Dixon, 1998) and (Ferguson, 1963) .

This paper will present the theory and experimental verification for a low cost method to predict the compressor performance under an alternative refrigerant based on existing compressor maps. A few factors that affect compressor performance maps, but whose effects cannot be predicted by similitude theory are also analyzed by reviewing large amount of tested data.

## 1. INTRODUCTION

Centrifugal compressors applied in commercial air conditioning compress refrigerant vapor to raise its pressure and temperature by converting kinetic energy into potential energy, i.e. pressure. Their pressure ratio or equivalently isentropic enthalpy rise mainly depends upon the saturation temperatures of condenser and evaporator which both are heat exchangers to exchange heat from/to refrigerant. Different geographic locations may indicate much different maximum temperature lift from evaporator to condenser, which requires centrifugal compressor to have either different size impellers or variable speed capability to provide the proper amount of refrigerant enthalpy rise. In addition, caused by environmental protection pressure, some environment friendly alternative refrigerants have been gradually replacing the current HCFC refrigerants. A lot of such applications are retrofit process (Brasz, 1996) which means the alternative refrigerant is just simply dropped into centrifugal chillers designed for old HCFC refrigerants. This requires accurate performance prediction of the centrifugal compressors based on their HCFC performance maps.

The performance map of the centrifugal compressor refers to the specified compressor discharge energy with given fluid path geometry when it passes a specific flow rate. Specific work delivered from rotor to operating gas can be given by Euler turbine equation:

$$w = U_2 V_{\theta 2} - U_1 V_{\theta 1} \quad (1)$$

From Euler equation different head can be reached through adjusting the amount of inlet pre-swirl. The inlet guide vane (IGV) thus is typically employed for a compressor stage to set proper pre-swirl. Therefore on compressor head-flow and efficiency-flow maps, It shows a 2D zone rather than a single line to reflect variable geometry characteristic. (Brasz, 2008 )

Usually the compressor performance characteristics are expressed as dimensionless variables employing the flow similitude theory to minimize the cost for lab test. Both (Dixon, 1998) and (Ferguson, 1963) gave the theory about the dimensionless maps as below:

$$\frac{\Delta h_{0s}}{(ND)^2}, \eta = f \left\{ \frac{\dot{m}}{\rho_{01} ND^3}, \frac{\rho_{01} ND^2}{\mu}, \gamma, \frac{ND}{a_{01}} \right\} \quad (2)$$

The definitions are described as:

Head coefficient or pressure coefficient: 
$$\psi = \frac{\Delta h_{0s}}{(ND)^2} \quad (3)$$

Flow coefficient: 
$$\phi = \frac{\dot{m}}{\rho_{01} ND^3} \quad (4)$$

Reynolds number: 
$$Re = \frac{\rho_{01} ND^2}{\mu} \quad (5)$$

Impeller blade Mach number: 
$$M_b = \frac{ND}{a_{01}} \quad (6)$$

Here  $\mu$ ,  $\rho_{01}$  and  $a_{01}$  are viscosity, gas density and speed of sound at stage inlet, respectively;  $N$  is impeller shaft speed usually measured as revolution per minute (RPM);  $D$  is impeller blade diameter;  $\Delta h_{0s}$  is the isentropic enthalpy rise of compressor stage;  $\gamma$  is the isentropic exponent and  $\dot{m}$  is the mass flow rate.

However similitude in compressible flow does not forecast the extent to which performance depends upon Reynolds number, isentropic exponent ( $\gamma$ ) and blade (tip) Mach number. Conclusions can only be drawn from experimental data (Dixon, 1998) and (Ferguson, 1963). Moreover these theories do not explain compressor stage working at irregular status such as during choke and surge. This paper investigates various factors that affect the compressor maps based on both similitude and the large amount of experimental data which were tested by Ingersoll Rand (IR) climate solution division (Trane). The prediction method is also provided to predict the performance of untested gas from available tested data.

## 2. EXPERIMENTAL APPARATUS

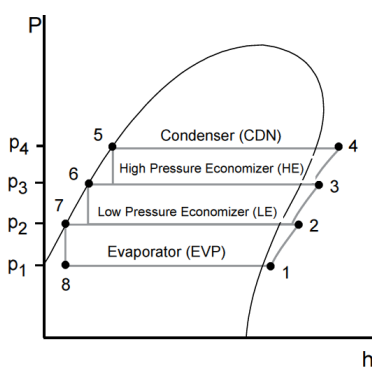
The tests were conducted on CenTraVac units, CVHG1100/CVHH1550, CVHF1300/CVHH1700, and CVHF870/CVHH1200 which are Ingersoll Rand-Trane's water cooled chiller products. The centrifugal compressor of CVHG1100/CVHH1550 chiller has three stages, works at 50 Hz commercial electricity while two stages, 60 Hz for the other two chillers. All chillers are directly motor driven. The nominal impeller shaft thus are 3000 RPM and 3600 RPM for 50 Hz and 60 Hz, respectively. In addition an Adjustable Frequency Drive (AFD) was also installed to drive the motor, which enables compressors to work at variable speeds. The operating gas passing through compressor are refrigerant R123, R245fa, FLUID-A and FLUID-B. (FLUID-A and FLUID-B are Ingersoll Rand internal names standing for Experimental Refrigerant A and B.) A series of impellers which possess same throat area and inlet and exit blade angle designs, but different blade diameters are tested for each unit as well. Some tests also consider different IGV openings, in which 90° stands for full opening or no inlet pre-swirl. The experiment matrix is listed in Table 1.

The experimental schematics and setups are shown in Figure 1a and 1b. The compressor intakes refrigerant vapor from evaporator (EVP), a big refrigerant shell with brass tube heat exchanger, where water passes through and rejects its heat to the refrigerant. After the first stage compression, the refrigerant vapor merges with the flow from low pressure economizer (LE, for 3-stage compressor shown in Figure 1a) or economizer (ECO, for 2-stage compressor shown in Figure 1b) that is a small flash tank before entering to next stage. Similarly, for the 3-stage compressor, after the second stage compression, the refrigerant vapor merges with the flow from high pressure

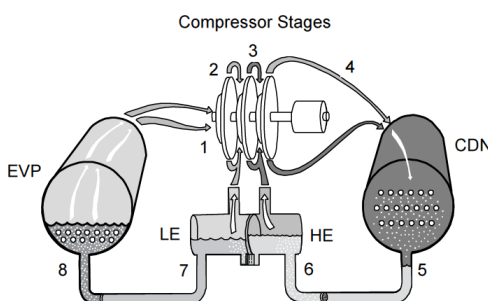
economizer(HE) that is, again, a small flash tank, shown in Figure 1a. The condenser (CDN), a big refrigerant shell similar to the evaporator, collects all compressed refrigerant vapor discharged from compressor volute conical exhauster, shown in both Figure 1a and 1b. Compressor head and flow can be adjusted by controlling water flow rate and temperature in both evaporator and condenser tubes. At each station indicated in both Figure 1a and 1b, five or six groups of total temperature, total pressure and static pressure probes are installed around the circumference. Their acquisitions, together with the refrigerant property definitions, are used to calculate the compressor performance maps. In this paper only the first stage performances are discussed, since the inlet flow of late stages are always messed up by the merging flow from economizers, which causes the results to be not typical.

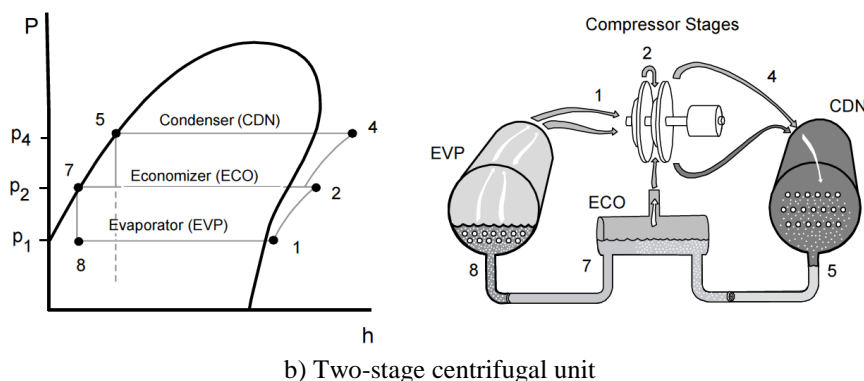
**Table1:** Experiment matrix

Test ID	Unit	Impeller Dia. (")	Gas	RPM	Test ID	Unit	Impeller Dia. (")	Gas	RPM
A-R2D34-30	CVHG1100/CVHH1550	34.25	FLUID-A	3000	C-R2D34-36	CVHF1300/CVHH1700	34.25	FLUID-A	3600
A-R2D34-27				2700	C-R2D34-33				3300
A-R2D34-23				2300	C-R2D34-30				3000
A-R3D34-30		34.25	R245fa	3000	C-R2D34-27		34	R123	2700
A-R3D34-27				2700	C-R2D34-23				2300
A-R3D34-23				2300	C-R1D34-36		31	R123	3600
A-R1D34-30		34	R123	3000	C-R1D34-33				3300
A-R1D34-27				2700	C-R1D34-30				3000
A-R1D34-23				2300	C-R1D34-27		28.25	R123	2700
A-R1D32-30		32.5	R123	3000	C-R1D34-23				2300
A-R1D29-30		29	R123	3000	C-R1D31-36				3600
A-R1D27-31		27	R123	3000	C-R1D28-36				3600
B-R1D34-36	CVHF870/CVHH1200	34	R123	3600	B-R1D30-30	CVHF870/CVHH1200	30	R123	3000
B-R1D34-33			R123	3300	B-R1D30-27			R123	2700
B-R1D34-30			R123	3000	B-R1D30-23			R123	2300
B-R1D34-27			R123	2700	B-R1D28-36		28.25	R123	3600
B-R1D34-23			R123	2300	B-R1D28-33			R123	3300
B-R4D34-36		34	FLUID-B	3600	B-R1D28-30			R123	3000
B-R1D30-36		30	R123	3600	B-R1D28-27			R123	2700
B-R1D30-33			R123	3300	B-R1D28-23			R123	2300



a) Three-stage centrifugal unit





**Figure 1:** Experimental schematics and setups (Ingersoll Rand, 2008)

### 3. RESULTS AND DISCUSSIONS

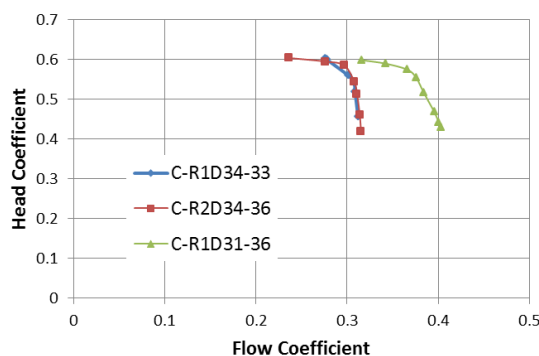
#### 3.1 Effects of isentropic exponent ( $\gamma$ ) and Reynolds number

Isentropic exponent ( $\gamma$ ) is a gas property and defined as the ratio between specific heat at constant pressure and that of at constant volume (Dixon, 1998) and (Brasz 1996). At inlet condition, typically 38°F for our tests, the values for isentropic exponent are 1.1028, 1.1047, 1.1004 for R123, R245fa, and FLUID-A respectively. The difference of the  $\gamma$  value is too small and its effect on the aerodynamics performance is hard to be detected.

Reynolds number can be calculated by equation (5) using stage inlet conditions. For example if a 34 inch diameter impeller rotating at 3600 RPM intakes 38°F saturated R123 vapor, the Reynolds number is  $8.19 \times 10^5$  given by equation (5). Take this case as bench mark. The Reynolds number ratio is defined as the Reynolds number for the specific case divided by the Reynolds number of this bench mark. In Table 2 three tests are listed. They are selected as with close blade Mach number in order to minimize its effect. Test C-R1D34-33 and C-R1D31-36 are on R123 and C-R2D34-36 is on FLUID-A. Blade Mach number calculated by equation (6) for the test C-R1D34-33 and C-R2D34-36 are 1.185 and 1.2, respectively. But the difference of Reynolds number ratio, 0.92 and 1.25 respectively, is about 36%. This big difference is not reflected on the map shown in Figure 2. Their curves are almost identical. On the other hand, the blade Mach number of test C-R1D31-36 that is on R123 also is 1.18, almost same as that of test C-R1D34-33. The Reynolds number ratio is 0.91, almost same too. There exists a big difference between these two curves on the map. This deviation is indeed caused by different impeller diameters, as discuss in late sections. Reynolds number does not affect the map very much or its effect is too small to be detected by the experimental apparatus. [Dixon] pointed out when the compressor operates at the fluid with high Reynolds number, its effect can be dropped. The Reynolds number for the refrigerants used in this test are all about ( $10^6$ ), therefore test results cannot reflect the explicit difference caused by Reynolds number effects.

**Table2:** Experiment summary for the study of Reynolds number effect

Test ID	Blade Mach#	Reynolds Number Ratio
C-R1D34-33	1.185	0.92
C-R2D34-36	1.2	1.25
C-R1D31-36	1.18	0.91



**Figure 2:** Maps at IGV=90° for the study of Reynolds number effect

### 3.2 Blade Mach Number Effect

Blade Mach number calculated by equation (6) describes how fast the blade tip linear velocity is. Obviously it affects the compressor performance map very much. Our tests show that there exists a range of blade Mach number within which the performance does not have much effect. This range can be as big as 0.2. Figure 3 shows three tests, C-R2D34-30, C-R1D34-30 and C-R1D34-27, that are done on CVHF1300/CVHH1700 unit and whose summary can be found in Table 3. The blade Mach number ranges from 0.97 to 1.077, while on the head coefficient-flow coefficient map, their curves are almost identical.

However shown in Figure 3, the curve for C-R1D28-36 whose blade Mach number, 1.074, is within this range shifted towards increasing of flow coefficient by about 45%. Moreover the curve is flatter, i.e. less head coefficient achieved on surge end. This is from the relatively small impeller used in the test. See Table 1 for test parameters. If the blade Mach number is same, the small impeller must rotate fast and thus to allow more flow passing through it. The impeller throat areas are same in our tests. Therefore the maximum volumetric flow rate passing through impellers should be same or independent of the impeller blade diameters at specified shaft RPM. The flow coefficient defined as volumetric flow rate divided by the cubic power of impeller blade diameter, see equation 4, is not applied to the situation that involves different impeller size. Figure 3b is same data as those in Figure 3a but plotted as head coefficient –Q/N (Volumetric flow/RPM), which clearly show the flow capacity for C-R1D28-36 is close to other tests. Similar situation can also be found in Figure 5a and 5b, same data plotted with respect to different non-dimensional flow parameters. Figure 5b plotted as Head coefficient- Q/N reflects clearer centrifugal compressor characteristics than Figure 5a plotted as head coefficient –flow coefficient. In this paper the unit for Q/N is [CFM/RPM], unless otherwise specified.

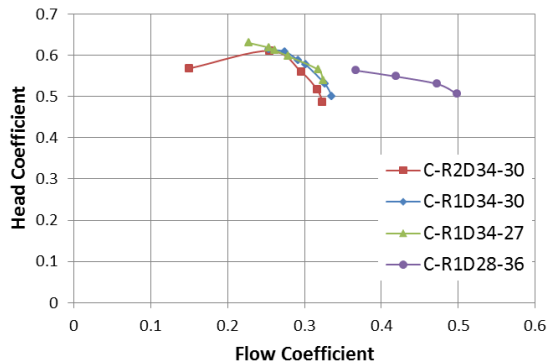
If the blade Mach number is high, the head drops fast when the flow increases to near choke limit. Or the choke characteristic becomes more apparently, see Figure 4 a, b, and d. For Ingersoll-Rand centrifugal compressors choke usually occurs in impeller throat area where the local gas velocity exceeds the speed of sound. Therefore the maximum volumetric flow passing through the throat is proportional to the speed of sound. All tested impellers for each unit are with same blade designs and thus same throat area. This provides designers an effective way to predict the aero performance of different operating gas based on the current data.

Figure 4 a shows two tests done on the CVHF1300/CVHH1700 unit. Impellers for both tests are 34" and rotate at 3600 RPM. The operating gases are FLUID-A (C-R2D34-36) and R123 (C-R1D34-36), respectively. At inlet, 38°F, the ratio of speed of sound from FLUID-A to R123 is 1.08. Multiplying its flow by this ratio, the C-R1D34-36 curve is shifted to obtain the prediction curve, shown in Figure 4 a. The predicted FLUID-A curve is almost identical with the tested one except the head is a little bit low on rotating stall/surge end. This is because the impellers are polished in FLUID-A test while as cast in R123 test and thus the isentropic efficiency for the former is higher, shown on Figure 4 c which shows the ratio of total-to-total isentropic efficiencies with respect to the peak efficiency of R123 performance.

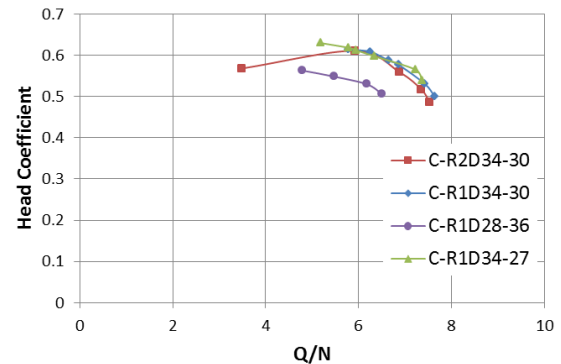
Figure 4 b show two tests whose configurations are exactly same as those tests shown in Figure 4 a except the shaft speeds are 3300 RPM. Multiplying the flow of R123 curve (C-R1D34-33) by the ratio of speed of sound, 1.08, one gets the FLUID-A prediction curve that is again close to the FLUID-A tested performance (C-R2D34-33).

**Table3:** Experiment summary for the study of blade Mach number effect

Test ID	Blade Mach#	Test ID	Blade Mach#
C-R2D34-30	1	C-R1D28-36	1.074
C-R1D34-30	1.077	C-R1D34-27	0.97



a) Plotted as head coefficient v.s. flow coefficient

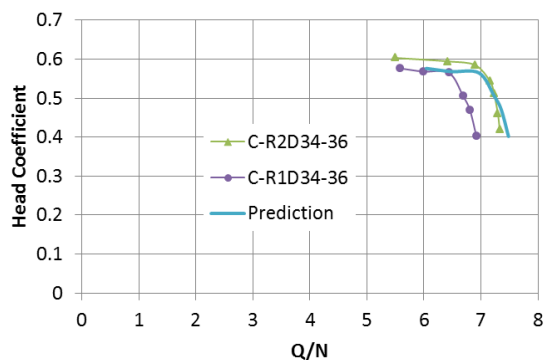


b) Plotted as head coefficient v.s.  $Q/N$

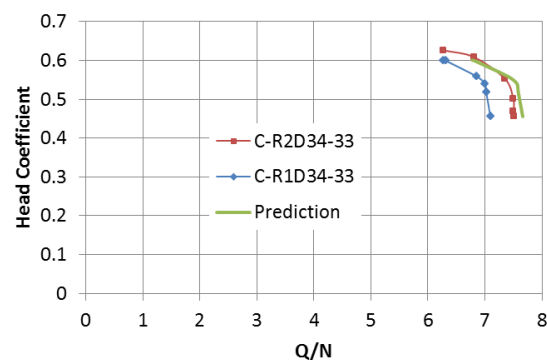
**Figure 3:** Map comparisons at  $IGV = 90^\circ$  for blade Mach number effect study

This can be further validated by tests done on CVHF870/CVHH1200 unit shown in Figure 4 d. The impellers of both tests are still 34 inch but different design compared to tests on CVHF1300/CVHH1700. Both tests are in same shaft speed, 3600RPM. In addition different IGv openings are also considered. The operating gases are FLUID-B (B-R4D34-36) and R123 (B-R1D34-36). At inlet,  $38^\circ\text{F}$ , the ratio of speed of sound from FLUID-B to R123 is 0.957. The prediction curve on Figure 4 d is obtained via shifting B-R4D34-36 curve by multiplying its flow by the ratio, 0.957. The predicted map is almost identical with the tested R123 performance, even for partially opened IGv.

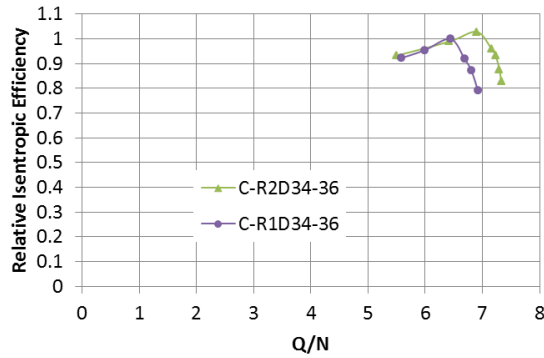
When its opening is low, the throttle effect of the Inlet Guide Vane becomes more apparent. There exists a big pressure drop across the IGv. Compressor impeller thus has to intake low pressure gas, which, from stage perspective, gives rise to a very low or even negative head coefficient. However compressor works properly. Cambio explained more details to explain this in literature (Cambio, 2000).



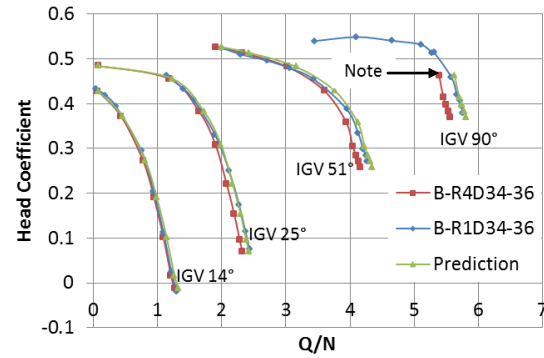
a) Prediction at  $IGV=90^\circ$  and nominal RPM



b) Prediction at  $IGV=90^\circ$  and reduced RPM



c) Efficiency Map at IGV=90° and nominal RPM

d) Prediction at various IGV and nominal RPM  
(Note: test was forced to stop because of AFD power limit )**Figure 4:** Performance prediction for tests with high blade mach number.

### 3.3 Impeller Size Effect

Impeller blade diameter  $D$  alone is another factor to affect non-dimensional performance maps. The impeller diameter decides the pattern of the head-flow curves and the choke characteristics of map curves. Even with same blade Mach number, different blade diameter will result in different curves on performance maps.

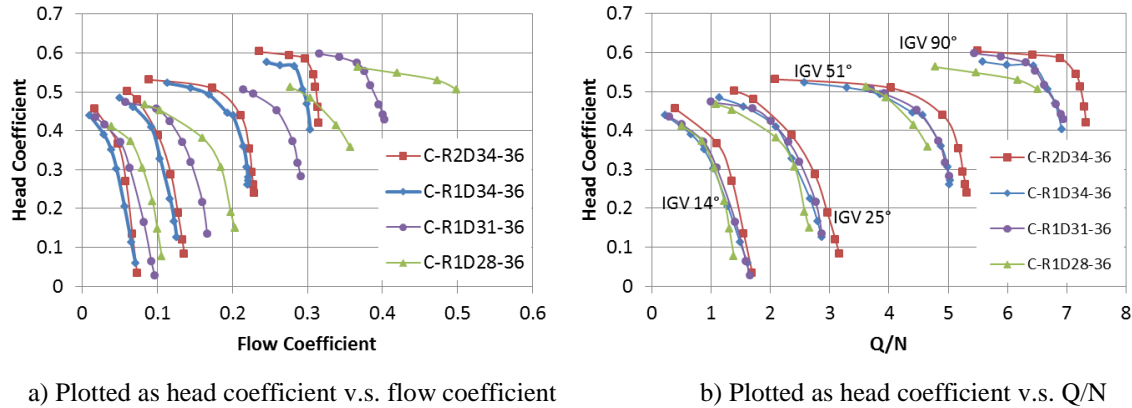
**Table4:** Experiment summary for the study of impeller size effect

Test ID	Blade Mach#	Test ID	Blade Mach#
C-R2D34-36	1.2	C-R1D31-36	1.18
C-R1D34-36	1.29	C-R1D28-36	1.07

Table 4 is the summary of the tests done on CVHF1300/CVHH1700 unit and their results are shown in Figure 5. Figure 5a and 5b show same data but in different non-dimensional flow parameters. The shaft speeds for all tests in Figure 5 are 3600RPM. Figure 6 shows the results tested on CVHF870/CVHH1200 unit with various impeller sizes and shaft speeds.

When the IGV is fully opened, the big impeller performance is more likely to go to rotating stall, which, shown on the map, losses the head when the flow decreases. While the medium and small impeller have much better surge performances. From Figure 5b, comparing C-R1D34-36 and C-R1D31-36, in spite of the difference of their blade Mach number shown in Table 4, both tests have almost identical map curves except at surge end of 90° IGV opening where the head of 34 inch impeller is lower. Similar head loss for the big impeller also seen in other shaft speeds, shown in Figure 6 a-d. Apparently this is nothing related to shaft speed, which, in turn, proves that it is caused by impeller size only.

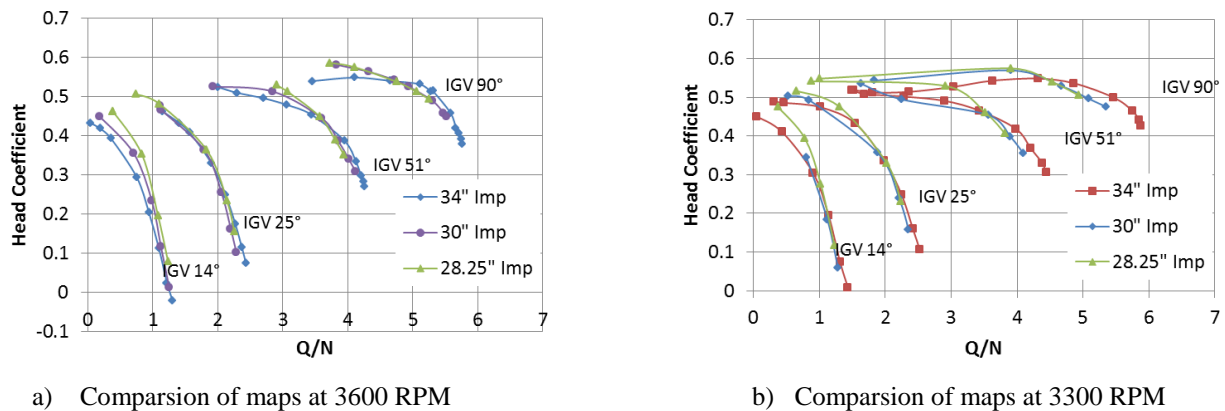


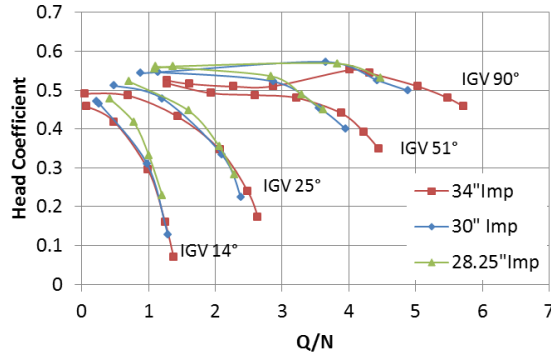


**Figure 5:** Experimental results for the study of impeller size effects

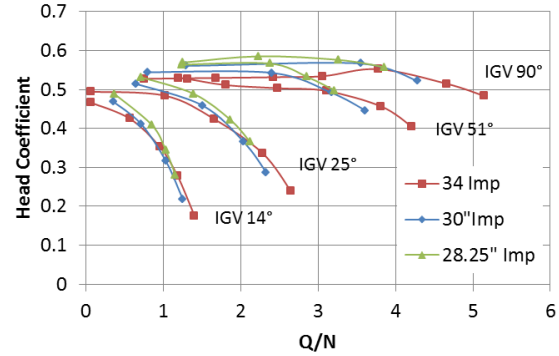
Figure 5 and 6 also indicate that the head coefficients are almost identical if the IGW opening is same, regardless the impeller size and shaft speed. For example shown in Figure 6, impellers operating at various shaft speeds, from 2700 RPM to 3600 RPM, the head difference caused by blade Mach number is not obvious. This is because head coefficient is defined as the isentropic enthalpy rise that is decided by Euler equation divided by square power of shaft speed times impeller diameter. Or the head coefficient is normalized to both impeller size and shaft speed.

However impeller size does affect maximum flow capability too. The big impeller usually have big choke limit. The maximum Q/N at specified IGW setting for big impeller is more than that of small impeller. This is independent of the blade Mach number and is only related to the impeller size. Both Figure 5 and Figure 6 reflect this fact, i.e. no matter what shaft RPM are the tests, the big impeller has more flow capacity when it hits choke limit. C-R2D34-36 in Figure 5 is different since its operating gas is FLUID-A. The effect of the blade Mach number causes it to have much greater flow capacity than other 34 inch impeller. This has already been discussed in section 3.2.





c) Comparison of maps at 3000 RPM

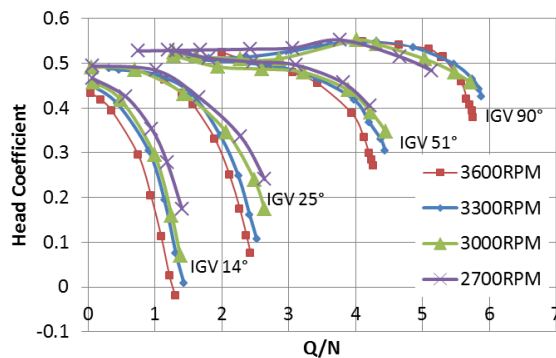


d) Comparison of maps at 2700 RPM

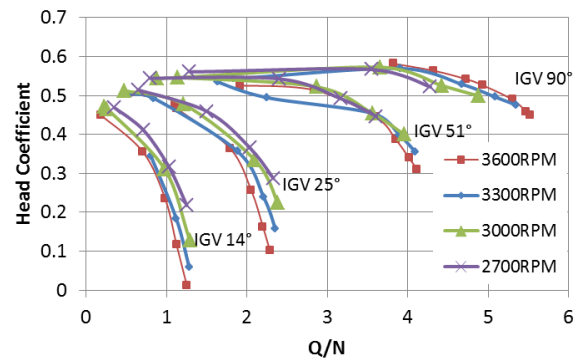
**Figure 6:** Impeller size effect at different shaft speeds. Maps are tested on CVHF870/CVHH1200 compressor and R123 fluid

### 3.4 Shaft Speed Effect

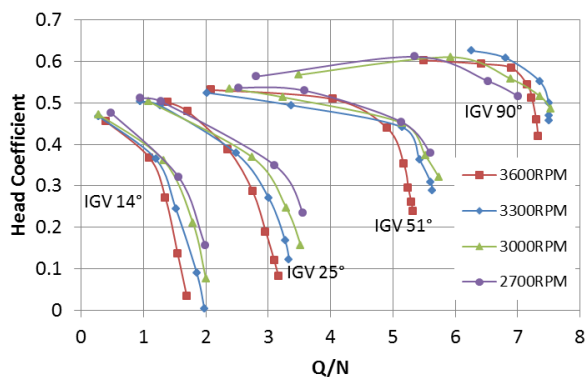
Impeller shaft speed is directly proportional to the blade Mach number whose effects is described earlier. Shaft speed alone has its own effect on the non-dimensional maps of stage aero performance. Figure 7 a-g show the performance map of tests done on CVHF1300/CVHH1700, CVHF870/CVHH1200 and CVHG1100/CVHH1550. The tests of studying shaft speed effect, Figure 7, are done in various operating gases and impeller sizes. The main effect of the shaft speed alone is on the partial IGV performance. The faster the shaft speed is, the more incidence loss caused by mismatch between flow direction and IGV blade camber line. This can explain why the head are lower for the high RPM test than that of low RPM ones, at the given IGV opening shown on all tests. This head loss is not related the impeller size and is also existed on medium and small impeller performance maps, shown in Figure 7 b, and d. Moreover this is not related to operating gas, or blade Mach number, either, see Figure 7 c, e-g.



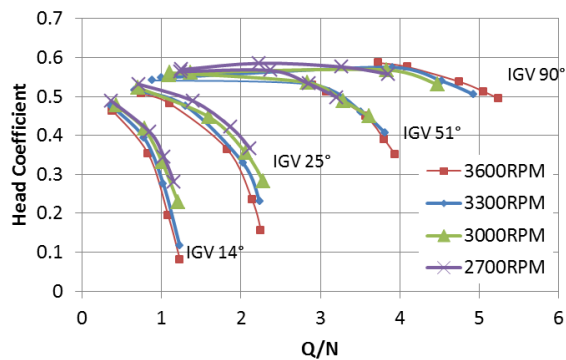
a) CVHF870/CVHH1200 compressor 34" impeller and R123



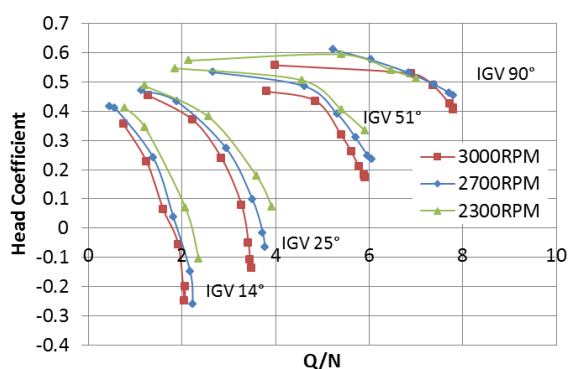
b) CVHF870/CVHH1200 compressor 30" impeller and R123



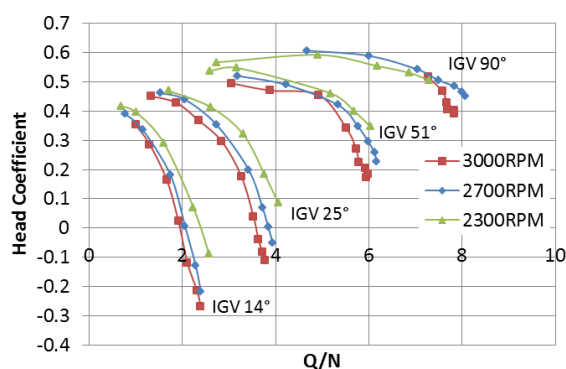
c) CVHF1300/CVHH1700 compressor  
34.25" impeller and Fluid-A



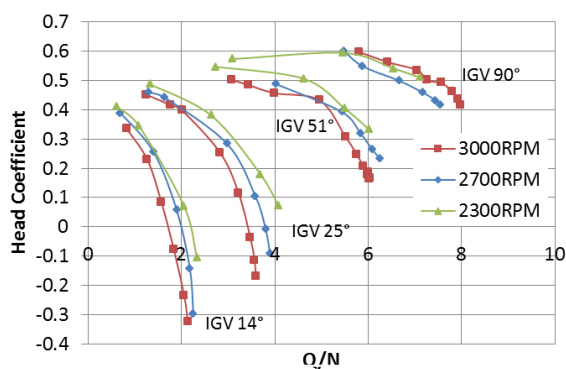
d) CVHF870/CVHH1200 compressor  
28.25" impeller and R123



e) CVHG1100/CVHH1550 compressor  
34.25" impeller and Fluid-A



f) CVHG1100/CVHH1550 compressor  
34" impeller and R123



g) CVHG1100/CVHH1550 compressor  
34.25" impeller and R245fa

**Figure 7:** Impeller shaft speed effect at various impeller sizes and operating gases

## 4. CONCLUSIONS

The non-dimensional performance maps for centrifugal compressor stages are affected by many factors such as fluid properties of operating gas, impeller blade tip Mach numbers, impeller size and shaft speed.

The values of isentropic exponent for low pressure refrigerants, such as R123, R245fa, FLUID-A and FLUID-B are almost identical and thus do not show much effect on compressor maps. The effect of unit related fluid property, Reynolds number, is negligible too, since all tested low pressure refrigerant vapors are with high Reynolds number.

Impeller blade Mach number affects performance map. But there exists a value rang within which blade Mach number shows limited effect on performance maps. In addition blade Mach number also decides stage choke character. It is possible to predict the performance for unknown operating gas based on the values of speed of sound when the blade Mach number is high.

Impeller blade diameter alone affects the performance maps too. The size of impeller decides pattern of map curves and flow capacity. The impeller shaft speed alone shows much effect on partially opened IGV performance maps, indicated by incident loss on IGV blades.

## NOMENCLATURE

$a$	Speed of sound	(foot/sec)
$D$	Impeller blade tip diameter	(inch)
$\dot{m}$	Mass flow rate	(lbm/sec)
$M_b$	Impeller blade Mach number	(-)
$N$	Impeller shaft speed	(RPM)
$Q$	Volumetric flow rate	(foot <sup>3</sup> /sec)
$Re$	Reynolds number	(-)
$U$	Impeller tangential velocity	(foot/sec)
$V$	Impeller absolute velocity	(foot/sec)
$w$	Euler work	(foot <sup>2</sup> /sec <sup>2</sup> )
$\gamma$	Isentropic exponent	(-)
$\Delta h$	Enthalpy rise	(BTU/lbm)
$\eta$	Isentropic efficiency	(-)
$\mu$	Gas viscosity	(lbm/(foot*sec))
$\rho$	Gas density	(lbm/foot <sup>3</sup> )
$\phi$	Flow coefficient	(-)
$\psi$	Head coefficient	(-)

### Subscript

0	Stagnation condition
1	Location of centrifugal compressor stage inlet
2	Location of centrifugal compressor stage exit
s	Isentropic condition
$\theta$	Tangential direction

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